DESIGN AND ANALYSIS OF IMPACT CRUSHERS

A THESIS SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS FOR THE DEGREE OF

Bachelor of Technology

in

Mechanical Engineering

By

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Department of Mechanical Engineering

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Rourkela

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Under the guidance of
Prof. N.Kavi

NATIONAL INSTITUTE OF TECHNOLOGY
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Department of Mechanical Engineering
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2007

CERTIFICATE

This is to certify that the project work entitled “Design and Analysis of Impact Crushers” by Sidhartha Patnaik and Biswajit Pattnaik has been carried out under my supervision in partial fulfillment of the requirements for the degree of Bachelor of Technology during session 2006-07 in the Department of Mechanical Engineering , National Institute of Technology, Rourkela and this work has not been submitted elsewhere for a degree.

Place: Rourkela
Date: 1/5/2007

Prof. N.Kavi
Professor
Mechanical Engg. Dept.
Rourkela-8

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We would like to take the chance to express our appreciation to our family members.Their continuous love and support gave us the strength for pursuing our dream.

Special thanks to our friends and other members of the department for being so supportive and helpful in every possible way.

NIT Rourkela

1st May,2007

Sidhartha Patnaik

Biswajit Pattnaik

ABSTRACT

Crushers are major size reduction equipment used in mechanical , metallurgical and allied industries.They are available in various sizes and capacities ranging from 0.2 ton/hr to 50 ton/hr.They are classified based on different factors like product size and mechanism used.Based on the mechanism used crushers are of three types namely Cone crusher,Jaw crusher and Impact crusher. Our objective is to design various components of an Impact crusher like drive mechanism, shaft, rotor ,hammers, casing ,feed and discharge mechanism which will be useful in minimizing weight, cost and maximizing the capacity. Impact crushers: they involve the use of impact rather than pressure to crush materials. The material is contained within a cage, with openings on the bottom, end or side of the desired size to allow pulverized material to escape. This type of crusher is usually used with soft material such as coal, seeds or soft metallic ores. The mechanism applied here is of Impact loading which is applicable when the time of application of force is less than the natural frequency of vibration of the body.

As because here the hammers are rotating at a very high speed, time for which the particles come in contact with the hammers is very small, so here impact loading is applied.

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CHAPTER:1

INTRODUCTION

Introduction

A crusher is a machine designed to reduce large solid chunks of raw material into smaller chunks.

Crushers are commonly classified by the degree to which they fragment not starting material with wares crushers not reducing it by much, intermediate cruiser fragmenting it much more significantly and grinders reducing it to a fine power.

Impact crushers: they involve the use of impact rather than pressure to crush materials. The material is contained within a cage, with openings on the bottom, end or side of the desired size to allow pulverized material to escape. This type of crusher is usually used with soft material such as coal, seeds or soft metallic ores:

- ❖ Hammer mills- Utilize heavy metal bars attached to the edges of horizontal rotating disks by hinges, which repeatedly strike the material to be crushed.
- ❖ Ball mills- Use metal balls in rotating cylinders.
- ❖ Stamp mills- Use cans to lift weighted vertical hammer which are dropped by gravity to crush the material.

DESIGN PARAMETERS:

The principal design parameters that drive crushing plant selection and configuration included.

1. Production Requirements
2. Ore characteristics
3. Project location
4. Operational considerations
5. Climate conditions
6. Capital cast
7. Safety & environment
8. Life of mine plants.
9. Maintenance requirements.

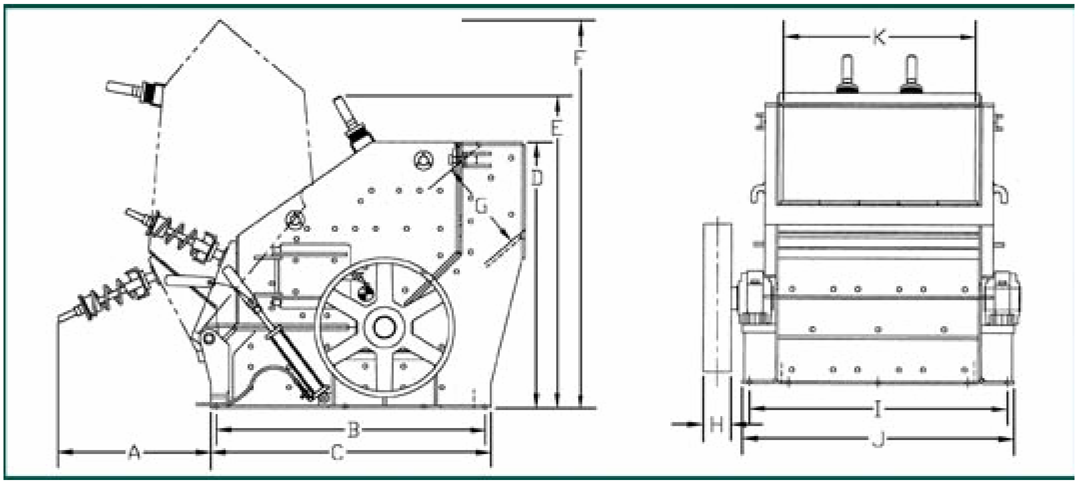


Fig.1

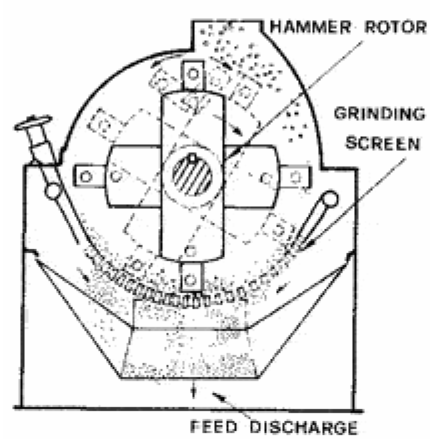


Fig.2

ADVANTAGES:

1. Hammers have four crushing positions to maintain a more const. gradation and greater top size control.
2. Less capital outlay.
3. High degree of product size control
4. Long life of wear components.

For different m/c wts the horse power required are as follows:

Wt (kg)	Horse power
15000	150
20000	200
25000	250
35000	300
45000	350
70000	400

The hammer is subjected to:

1. Shear force at point of fixation (attachment).

$$F = \frac{n\gamma w^2}{area\ \ of\ \ catet} < (\tau)all.$$

γ = distance from axis of shaft.

2. Centrifugal force on hammers mrww2

   Compressive force

$$F = \frac{m\gamma w^2}{face\ \ area}$$

3. Bending due to raw material strike

a) Depends on the non flow rate

Shaft is subjected to:

1. **Torsion**

Max. Torsion will be experiences at L/2.

L= length of shaft.

2. Bending

Assume it to be a cantilever of length L/2.

Principle of operation:

Feed material drops through the feed tube onto chi shoe table or enclosed rotor which, through centrifugal force throws the material against stationary anvils when the rock impacts the anvils at 900 angle, it shelters along natural grain structures creating a uniform cubical product. This is simple and economical to operate. Product output is easily controlled by varying the rotor speed. each crusher can accommodate several different rotor, shoe table anvil ring and auto generous rock shelf combinations.

Horse Power	---------	Model	----------	Capacity (Tonnes)
75-125				5-50
150-250				30-125
200-300				50-175
200-400				100-275
300-600				200-400
250-500				150-400
300-800				250-500
800-1000				600-1300

VERTICAL SHAFT IMPACT CRUSHER

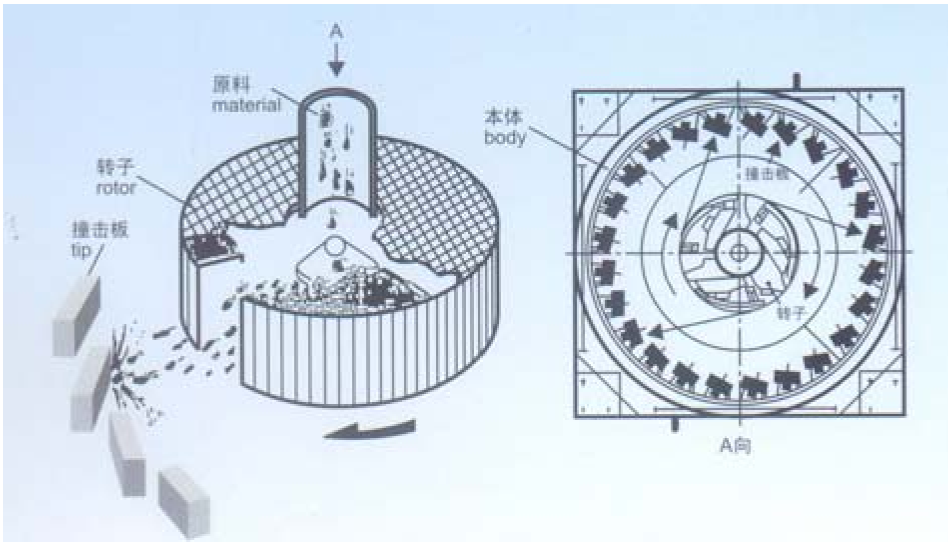


fig 1.3

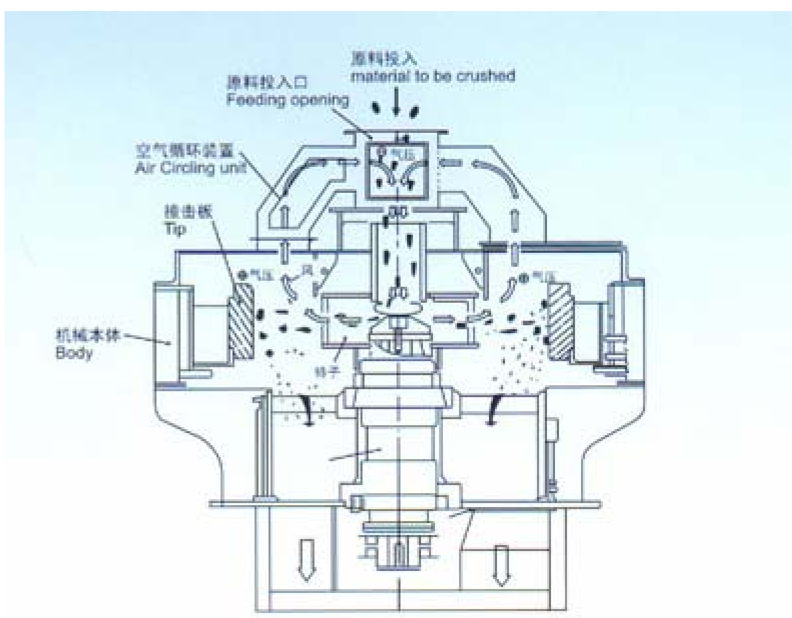


fig 1.4

CHAPTER 2

DESIGN AND CALCULATION

DESIGN OF HORIZONTAL SHAFT IMPACT CRUSHER

HORIZONTAL SHAFT IMPACT CRUSHER

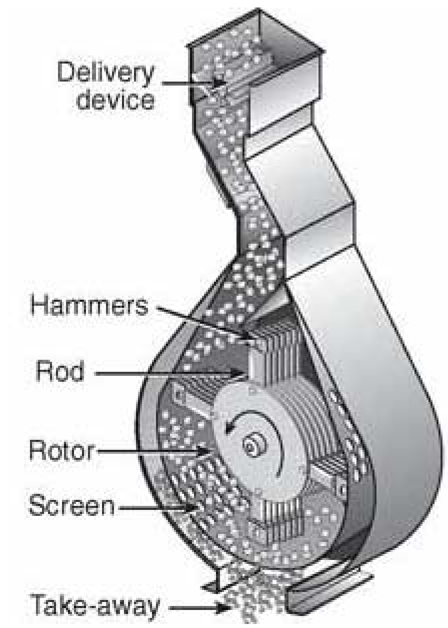
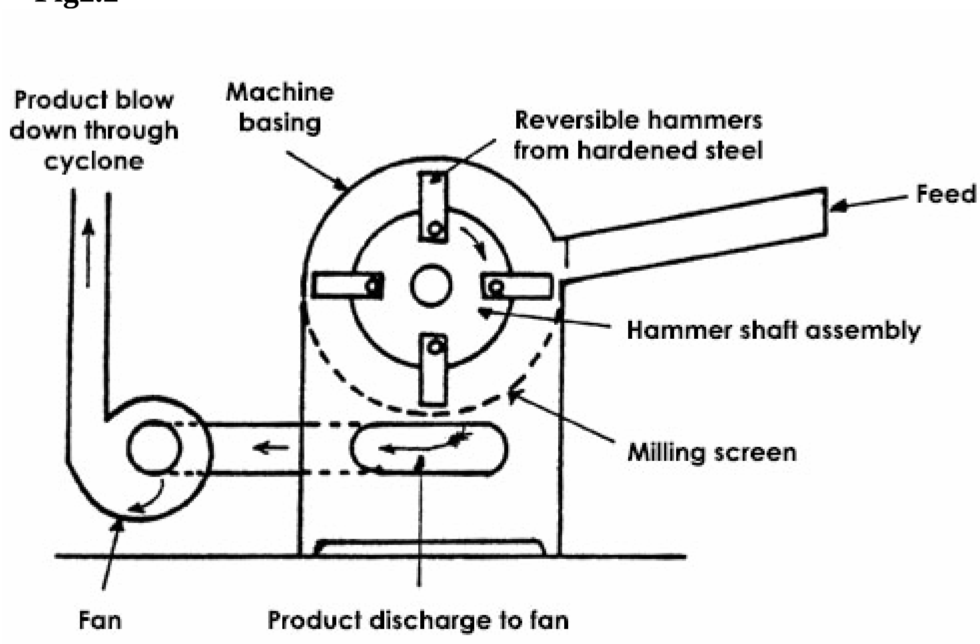


Fig2.1

Fig2.2



1. Design of V-belt Drive

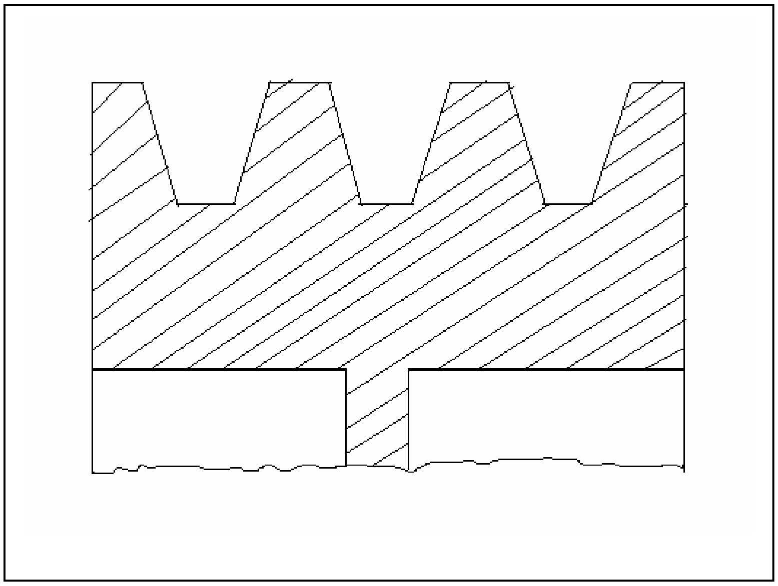


Fig.3

MATERIAL SELECTION: LEATHER

Mechanical properties:

$\mu = 0.15$

σ all=7 MPα=7N/mm^2

ρ= 1.2 x 103 Kg/m^3

Design calculations:

P = 15 HP

= 15 x 746 W $\approx$ 12 Kw

$2\beta = 30^0$ (assumed)

N = 1440 rpm

α = 150 mm

W =17 mm

T = 11 mm

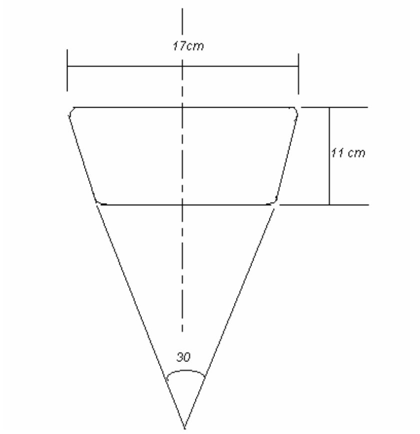


Fig.4

Speed ratio = 1.6

D = 220 mm

Tan 15^0 = 8.5 /b

B= 31.7 mm]

Tan 15^0 = P/20.7

2P = 11.1 mm

a = ½ (28.1) (11)= 154.5 $mm^2 \approx$ 160 mm^2

V = π x 0.15 x 1440 /60 = 11.3 m/s

Mass of belt /m = 0.19 kg / m

Tc = mv2 = o.19 x 11.3 2 =24.26 N

Cmin = 0.55 (D+d) + T

= 0.55 (150 +220) + 11

=214. 5 mm

C = 230 mm

θ =2 cos-1 D-d /2c = 144.560

$$\frac{T_1}{T_2} = e^{\mu\theta} \qquad \Rightarrow T_1 = 1.461 T_2$$

12 = (T1 – T2) x 11.3 x 2

$$\Rightarrow T_1 - T_2 = 0.531kn$$

$$0.461T_2 = 531 \qquad \Rightarrow T_2 = 1152N$$

$$T_1 = 1683N$$

$$1707 = \sigma \;\; x\,160\;mm^2$$

$$\Rightarrow \sigma = 10.67N/mm^2\sigma au$$

Design fails

$$\frac{11}{b} = \tan 15^0$$

$$\Rightarrow b = 41.05mm$$

$$\tan 15^0 = \frac{P}{41.05 - 14}$$

$$\Rightarrow 2p = 14.5mm \approx 15mm$$

$$a = \frac{1}{2}x(14)x37 = 260mm^2$$

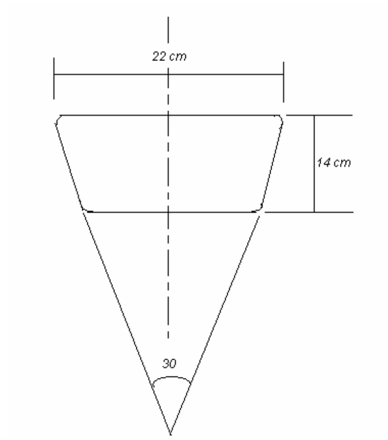


Fig.5

Cmin = 0.55 (370 +14

= 218 mm ≈230 mm

T = σ x a →σ = 1707/260 = 6.56 N/mm2 , < σ_{au}

Hence design is safe.

2- Shaft Design

Material of hammer: CI

Density of CI = 8000 kg /m^3

Cross Section of hammer

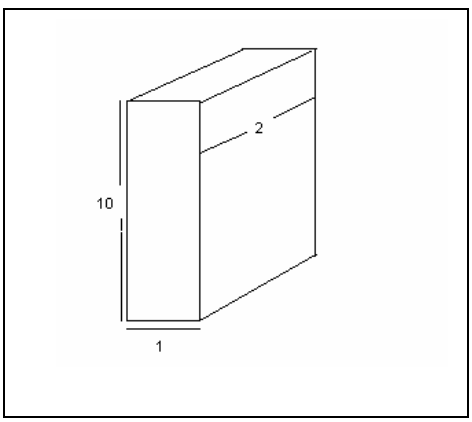


Fig.6

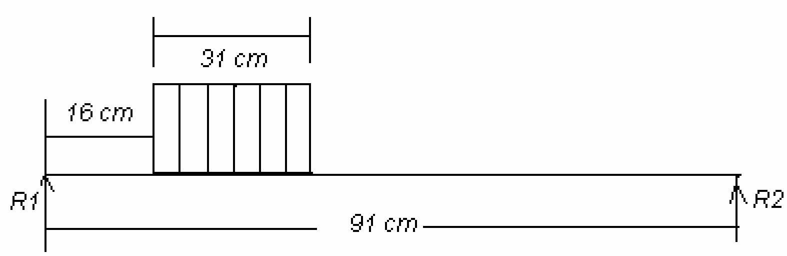


Fig.7

$N = 20" \times (2.54)^3\ cm^3$

$= 312.5\ cm^3$

$Wt = 6 \times 8000 \times 312.5$

$= 15.73\ kg$

$W = 1.96\ kg/m$

$R_1 + R_2 = 15.73 \times 9.8 = 154.154\ N$

$R_2 \times 91 = 154.73 \times 31.5$

$R_2 = 53.36\ N,\ R_1 = 100.6\ N$

$M = R_1x - w(x-16)(x-16/2)$

$= 0.98\ x^2 + 131.96x - 250.88$

For maximum bending moment

$dm/dx = 0$

$\Rightarrow 100.6 - 0.98(2x-32) = 0$

$\Rightarrow x = 67$

Maximum $= 100.6 \times 67 - 0.98(672 - 256 - 32 \times 67)$

$= 57639.2\ N\text{-}cm$

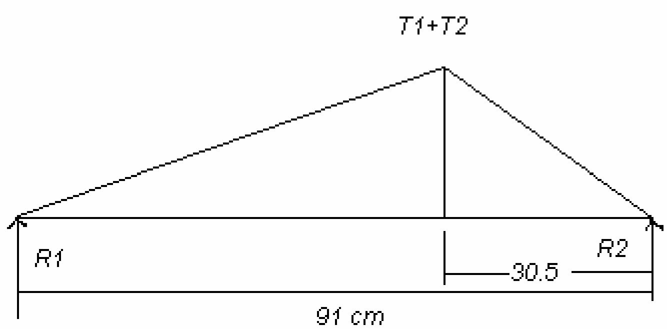


fig 8

Now considering the bending moment due to tension on both sides of the belt

$T_1 + T_2 = 2835 = R_1 + R_2$

- ⇨ $R_2 \times 91 = 2835 \times (60.5)$
- ⇨ $R_2 = 1884.8\ N$
- ⇨ $R_1 = 950.2\ N$
- ⇨ Maximum $= R_1 \times 60.5$

= 57486.63 N-cm

Now , $M = \sqrt{M_1^2 + M_2^2} = 87706.32N - cm$

$T = {}^{p}\!/_{w} = 12.24N.m = 122gN.cm$

We know,

$M_e = \frac{1}{2}\left[M + \sqrt{M^2} + T^2\right]$

$= \frac{1}{2}\left[81406.32 + \sqrt{(81406.32)^2} + (1224)^2\right]$

$= 81410.9 \approx 81411\ N - cm.$

$T_e = \sqrt{M^2 + T^2} = 81415.52N - cm.$

Material of shaft : EM 28

Mechanical Properties:

$\sigma ut = 618N/mm^2$

$\tau ut = 490N/mm^2$

$\sigma yp = 392N/mm^2$

$\tau yp = 147N/mm^2$

Calculation of FOS

For tension

$a = \sigma ut/\sigma yp = 1.58$

F.S = 1.58 x 1.2 x 1.2 = 2.8 ≈ 3

For shear

$a = \tau ut/\tau yp = 3.33$

F.S= 3.33 x 1.2 x 1.2 = 4.8 $\approx 5.$

Design:

$T_e = \frac{\pi}{16} d^3 \tau_s$ (Max. shear stress thing)

⇨ $81415.52 = \frac{\pi}{16} x(75)^3 x\tau_s$

$$\Rightarrow \quad \tau_s = 983.3627 N/cm^2$$

$$(\tau_s)_{all} = (\tau_s)_{ut} / F.S = \frac{490}{5} = 98 N/mm^2$$

$$= 98 \ x \ 10^4 \ N/cm^2$$

$\tau_s < (\tau_s)_{all}$ hence design is safe.

According to max principal stress theory

$$\sigma = \frac{32Me}{\pi d^3} = \frac{32x81411}{\pi x(7.5)^3}$$

$\sigma < \sigma$ all hence design is safe

3. Hammer design:

Impact loading

A load that is suddenly applied to a m/c or structure is called an impact load. The effect of impact loads differs appreciably from that of static loads because, with a suddenly applied load, both the magnitude of the stresses produced and resistance properties of the material are affected. The most common type of impact testing is the “notched bar testing” or the izode impact test.

Impact bending stress:

Cantilever bean subjected to concentrated load.

Assuming the open screen area / hammer

= 50 % of total area / hammer

$$= 10x2x\frac{50}{100}x2.54cm^2 = 64.52cm^2$$

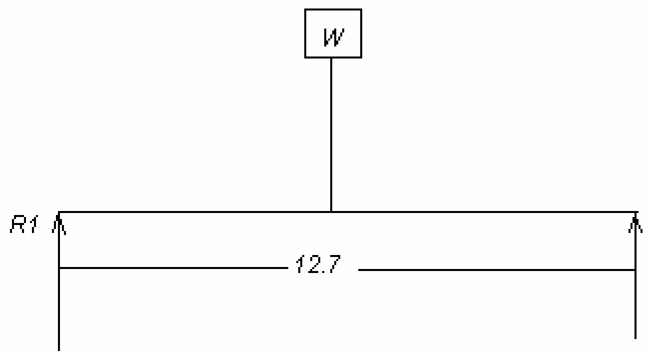


fig 9

Total tonnage = 2000 kg /hr

= 0.55 kg / sec

Tonnage / hammer = 0.55 /24

= 0.023 kg /sec.

Assuming it to be a concentrated coal falling from a distance 1 ft.

Applying impact equation

W(h+y) = py/2

Where P is the equivalent static load

⇨ 0.023 (30.5 +y) = py/2

Now for a simply supported bean subjected to a centre load the deflection is

$$Y = \frac{PL^3}{48EI} \qquad \Rightarrow P = \frac{48EI}{L^3}Y$$

Now, 0.023 v(305 + y) $= \frac{48EI}{2l^3} Y^2$

- ⇨ 0.0117EIY2 – 0.023y-0.7015=0
- ⇨ 3901440y2-0.023y-0.7015=0
- ⇨ Y=4.25 x 10 -4 cm

$$P = \frac{48EI}{l^3} y = 38902.64N$$

Max, Bending moment,

Maximum $= \frac{PL}{A}$

Max. stress produced.

$$S_{yp} = \frac{MaximumXc}{I} = \frac{\left(pL/A\right)xc}{I}$$

$$= 3952.57\ N/cm^2$$

Hence design is safe.

Impact Bending stress

Cantilever beam subjected to uniformly distributed load.

Assuming open screen area/ hammer

= 50% of total area/hammer

=10 x 2 x 50/100 x 2.54 cm^2

= 64.52 cm^2

a) Total tonnage = 2000kg /hr

= 0.55 kg /sec

b) Tonnage / Hammer = 0.55/24 = 0.023 kg /sec.

Assuming it to be uniformly distributed load taking from a height of 1ft.

W (h + y) = py /2.

The B.M at any section X distance x from the fixed and is given Lg,

$$EI = \frac{d^2y}{dx^2} = -\frac{w}{2}(l-x)^2$$

$$\text{Integrating} \quad EI\frac{dy}{dx} = \frac{w}{6}(l-x)^3 + C_1$$

$$x = o, \frac{dy}{dx} = o, \quad \text{EIy} = \text{-}\frac{\text{w}}{24}(l-x)^4 - \frac{wl^3}{6} + c_2$$

$$At \quad \text{x} = 0, \ \text{y} = 0$$

$$\rightarrow c_2 = wl^4 / 24$$

$$cEIy = -\frac{w}{24}(l-x)^4 - \frac{wl^3}{6}x + \frac{wl^4}{24}$$

work done due to impact distributed load,

$$= \frac{w}{l}dx\left[h - \frac{w}{24}\frac{(l-n)^4}{EI} - \frac{wl^3}{6EI}x - \frac{wl^4}{24EI}\right]$$

Net work done = $\int aw$

$$== \frac{w}{l}\int_o^i\left[hdx - \frac{w}{24}\frac{(i-x)^4}{EI} - \frac{wl^3}{6EI}x + \frac{wl^4}{24EI}\right]$$

$$= \frac{w}{l}\left[hi - \frac{wfe}{24EI}\frac{(l-x)^5}{5}\int_0^l - \frac{wl^3}{6EI}\frac{l^2}{z} + \frac{w}{24EI}\frac{l^5}{5}\right]$$

$$= \frac{w}{l}\left[hl + \frac{w}{24EI}\left(\frac{-l^5}{5}\right) - \frac{wl^5}{12EI} + \frac{wl^5}{120EI}\right]$$

$$= \frac{w}{L}\left[HL - \frac{wl^5}{12EI}\right]$$

$$= W\left[hl - \frac{wl^5}{12EI}\right]$$

h = 1ft = 12 inch = 30.5 cm

w = 0.023 kg/s. l = 12.7 cm

EI = .33298.51 mm^2

$$\text{W} = 0.023\ \left[\frac{30.5}{100} - \frac{0.023 \times (12.7)^4}{100^4 \times 12 \times 33298.51}\right]$$

= 0.007014996 J/s

static load work done

$$= \int^{c} \frac{p}{2lEI}\left(-\frac{w(l-x)^4}{24} - \frac{wl^3}{24}x + \frac{wl^4}{24}\right)dx$$

$$= \frac{p}{2lEI}\left[\frac{-w(-1)}{24}\left\{\frac{(l-x)^5}{5}\right\}_0^l - \frac{wl^3}{6}.\frac{l^2}{2} + \frac{wl^4}{24}l\right]$$

$$= \frac{p}{2lEI}\left[\frac{-wl^5}{120} - \frac{wl^5}{12} + \frac{wl^5}{24}\right]$$

$$= \frac{p}{2lEI}\left[\frac{-wl^5 - 10wl^5 + 5wl^5}{120}\right] = -\frac{p\,\text{wl}^5}{40lEI}$$

$$0.007014996 = \frac{P}{40\times\frac{12.7}{100}} \times \frac{1}{33298.51}\left[0.023\times\left(\frac{P.7^5}{100}\right)\right]$$

$$\rightarrow \text{P} = 33298.51 \text{ X } 0.4 \text{ X } 12.7 \text{ X } 0.007 \text{ X } \left(\frac{100}{12.7}\right)^5 \times 0.023$$

$$= 1184.01\times\left(\frac{100}{12.7}\right)^5 \times 0.023$$

$$\text{Mmax} = P\,{}^{l}\!/_{2} = 8.24\times10^5$$

$$\text{Mmax} = 8.24\times10^5 \times \frac{12.7}{200}$$

$$= 0.523\times10^5\ N.M$$

$$\sigma b = \frac{0.523\times10^5 \times 2.54}{100\times 27.75\times10^{-8}}$$

$$= 47.9\times10^8\ n/m^2$$

= 47.9 x 10^4 n/cm^2

= 0.47 NPa

considering the load to be acting on 1 hammer

$$\text{net workdone} = w\left[h - \frac{wl^4}{12EI}\right]$$

$$= 0.55\left[\frac{30.5}{100} - 0.55\times\left(\frac{12.7}{100}\right)^4 \times \frac{1}{12\times 33298.51}\right]$$

= 0.1677499 J/s

$$0.1677499 = \frac{p\times 0.55\times\left(12.7\!\left/100\right.\right)^5}{40\times\left(12.7\!\left/100\right.\right)\times 33298.51}$$

$$\rightarrow \text{P} = \frac{33298.51\times 0.1677499\times 40\times(100)^5}{(12.7)^5\times 40}$$

= 0.0183 x 10^{10} = 1.83 x 10^8

$$\text{Mmax} = 1.83\times 10^8 \times \frac{12.7}{200}$$

= 0.110 X 10^8 n-m

$$\sigma\, \text{b} = 0.116\times 10^8 \times \frac{2.54}{100\times 27.75\times 10^{-8}}$$

= 10636.42 x 10^8 N/m^2

= 106.36 mPa

Cantilever Beam Subjected to End concentrated Impact load.

w(h+y) = py/2

$$y = \frac{Pl^3}{3EI} \Rightarrow p = \frac{3EIY}{l^3}$$

$$w(hty) = \frac{3EI}{l^3}.\frac{y^2}{z}$$

$$\Rightarrow 0.55\left(\frac{30.5}{100} + y\right) = \frac{3\times 33298.51}{\left(12.7\!\left/1W\right.\right)^3}\frac{y^2}{z}$$

$$\Rightarrow 0.55(0.305ty) = 2.438399\times 10^7 y^2$$

$$\Rightarrow 0.16775 + 0.55y = 2.438399\times 10^7 y^2$$

$$\Rightarrow 2.438399\times 10^7 y^2 - 0.55y - 0.16775 = 0$$

$$y = 0.55 \pm \frac{\sqrt{(0.55)^2 + 4\times 2.438399\times 10^7 \times 0.16775}}{2\times 2.438399\times 10^7}$$

= 829.54 x 10^{-7} m

= 829.54 x 10^{-4} mm

= 0.08 mm or approximately equal to 0.1 mm

$$p = \frac{3EI}{l^3} y$$

$$= \frac{3 \times 33298.51}{(0.127)^3} \times 829.54 \times 10^{-7}$$

= 4045.5 N

Mmax = Pl = 4045.5 x 0.127

= 573.78 N.m

$$\Rightarrow 573.78 \times \frac{i}{I} = \sigma \text{ drawing}$$

c = 1” = 2.54 cm

$$\Rightarrow \sigma \text{ b} = \frac{573.78 \times 2.54}{100 \times 27.75 \times 10^{-8}}$$

= 0.47 x 10^8 N/m^2

= 0.47 X 107 N/cm^2

= 4700 N/cm^2

DESIGN OF HAMMERS CONSIDERING:

1. END LOADING WITH FATIGUE

Fatigue loading is not applicable in this case

2. By using Strain Energy method and approximating the loading to be a static one.

Shear stress at any distance y.

$$q = \frac{F}{Ib} \overline{Ay} \quad \text{F = Shear force}$$

$$= \frac{F}{Ib}(b)\left(d/2 - y\right)\left(d/4 + y/2\right)$$

$$\text{I} = \frac{bd^3}{12}$$

$$\Rightarrow q = \frac{6F}{bd^3}\left(\frac{d^2}{4} - y^2\right)$$

$$dv = \frac{q^2}{2G}(volume)$$

$$= \frac{1}{2G}\left[\frac{6F}{bd^3}\left(\frac{d^2}{4} - y^2\right)\right]^2 .(bdy)(dx)$$

Total shear strain energy

$$= \int_o^L \int_o^{d/2} dv$$

$$= \frac{1}{2G}\int_0^L\int_0^{\frac{d}{2}} {}^1\left(\frac{6F}{bd^3}\right)^2\left(\frac{d^2}{4} - y^2\right)^2 b\, dx.dy$$

$$= \frac{3}{5}\frac{F^2L}{bdG}$$

Workdone = w = ½ pys

Equating, $y_s = \frac{6}{5}\frac{F^2L}{bdG}$

⇨ $U = \frac{3}{5} \; \frac{P^2L}{bdG}$

$$Y_s = \frac{6}{5} \; \frac{PL}{bdG}$$

$$P = 0.55 \; \frac{kg}{sec} = 5.4N/\sec$$

$$G = 80N/mm^2$$

$$Y_s = \frac{6}{5}x\frac{5.4x(127mm)}{(2.54x2x2.54)(100)x80}$$

$$= 0.06mm$$

Material can be introduced by means of variable speed vein feeder. This type of feeder can have its motor slowed by a programmable controller to the main drive motor of the hammer mill.

CONVEYOR CALCULATION:

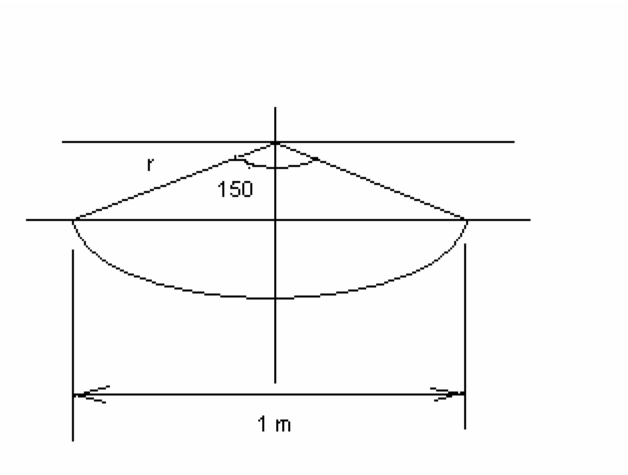


fig 10

Material: Rubber(1140 kg/m^3)

Angle: 15^o

Feed Rate: 2 metric ton/hr

We know,

Discharge(Q) = velocity(v) * Area(A)

= v*5/6*Π*r*(1 m)

⇨ r*v = 1/15* Π

Radius (r)	Velocity(v)
10 cm	2.1 cm/s
11 cm	1.9 cm/s
13 cm	1.6 cm/s
15 cm	1.4 cm/s

CHAPTER 3

A PERFORMANCE MODEL FOR IMPACT CRUSHER

ANALYSIS

A performance model for impact crushes (study)

The goal of the study is to predict the product size distribution, provided that the crushers rotor velocity and radius as well as the feed rate and size distribution are known before hand.

Classification and breakage Fn's

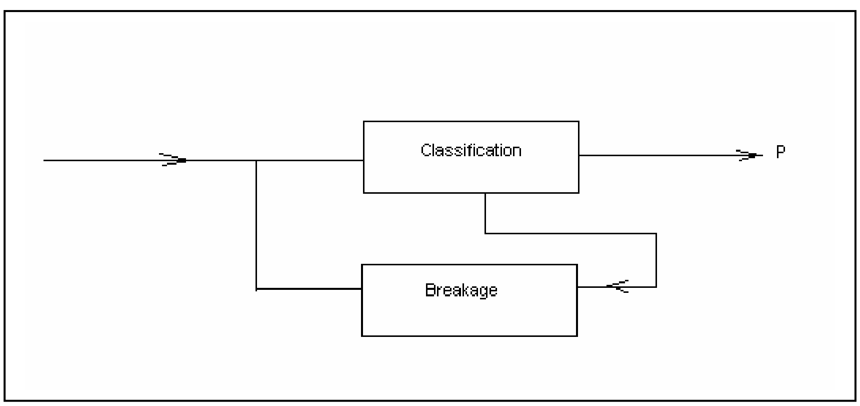


Fig 1

General scheme of breakage process

- ❖ Impact breakage takes place on a very short time scale and implies a dynamic crack propagation that leads to a much faster failure of particles.
- ❖ The impact generates compressive and tensile shock waves traveling throughout the particle. The presence of a significant, rapidly growing tensile stress helps the particles to break from within.

Mass Balance

F → feed

P → product

C → Classification operator, computes the probability of breakage of each particle size.

B → breakage operator

Governs the redistribution of broken particles in the preliminary defined size classes.

$P = (I-c), (I-B.C)^{-1}, f$

Where, I → identity matrix

IMPACT ENERGY PER UNIT MASS:

HORIZONTAL SHAFT:

Assumptions:

1. Rotor mass much greater than mass of single particles in the feed
2. Before impact, linear velocity of the crushing bar is much more important than the particle velocity. Hence KE of particles is negligible.

Considering the COLM,

$E = 0.5\ (R + 0.5Hb)^2.\ W^2$

Where, R → Rotor radius

Hb → height of impact surface of crushing bar.

W → rotor angular velocity

Vertical shaft

Assumption

1. particle energy does not change during its flight from the rotor periphery to the crushing walls.

$E = R_v^2.\ W^2$

Rv → rotor radius

W(S) → angular velocity

Classification function:

For cone and jaw crushers

$Co(DI) = 1 - [(d_i - k_2)/(k_1 - k_2)]$

$Ci(di) \rightarrow$ probability of breakage for a particle of size di (mm)

$K_1 \rightarrow$ min, size of particles that undergo breakage

$K_2 \rightarrow$ max. particle size found in product

$m \rightarrow$ shape parameter

but in this fn, K1 & K2 are static variables in impact fracture of particles, the prab of impact breakage depends mainly on its size and impact kinetic energy which is a dynamic variable.

So,

$C_i(d_i) = 1 - \exp[-C(d_i - d_{min})/d_{min})^k]$

Where,

d_{min} = min. size of particles that undergo breakage for the given operating condn's.

k = controls the shape of the classification fn.

BREAKAGE FUNCTION

The breakage distribution to bij represents the fraction of the debris created from breakage of identical parent particles of size dj and passing through a screen with mesh size di.

Bij (di, dj) = φ(di/dj)m + (1- φ).(di/dj)l

Φ = mass fraction of fine product

M,l = material co-efficient

The breakage matrix B for N screens of mesh sizes Di (i=1, N-1)

Bij = b(c-i)j (Di-1,dj) – bij (Di,dj)

Bjj= 1-bjj (Dj,dj)

Also di is the representative size of particles with dimension, where

Di>di>Di+1

RESULT

This model is able to predict the product size distribution with reasonable accuracy even when important variations in both the rotor velocity and feed are imposed.

CHAPTER 4

HAMMER LOCKING MODEL FOR IMPACT CRUSHER

HAMMER LOCKING ARRANGEMENT FOR IMPACT CRUSHER:

An impact crusher is disclosed having a rotor carrying a pair of hammer bars, with each hammer partly within an axially extending hammer partly within an axially extending peripheral scoot diametrically opposite each other. A portion of each leading hammer face within the slot and a facing wall of the slot co operate to define there between a generally cylindrical cavity parallel to an axis's about which the rotor rotates.

A two piece bushing is fitted in the cylindrical cavity at each end of the rotor, and each of the bushings comprise a pair of bushing halves each having a convex seoni cylindrical outer surface. The bushing halves are arranged within the cylindrical cavity with one convex surface engaging the hammer defined portion of the cavity and the other convex surface engaging the slot wall defined portion of the vacity, and a diametrical plans along which the hushing is split into halves, being substantially parallel to the adjacent leading hammer face. The halves of each bushing defined there between an internal conical shaped cavity with an apex end pointed inwardly of the adjacent end of the rotor. A rod passed through central bores in both plygs and nuts threaded on both rod ends holds the plugs relative to each other. Retaining walls are provided within the cylindrical cavity to limit movement of the bushings inwardly of their respective adjacent rotor ends.

Movements of the frusto conical plugs toward each other therefore moves each pair of bushing halves apart. As each pair of bushing halves move apart their convex outer semi cylindrical surfaces move apart and apply forces which are directed radially out of the bushings to both push the hammer radially into its slot and tangentially against the back wall of the slot, to securely lock hammer in its slot.

DESCRIPTION OF THE DRAWINGS

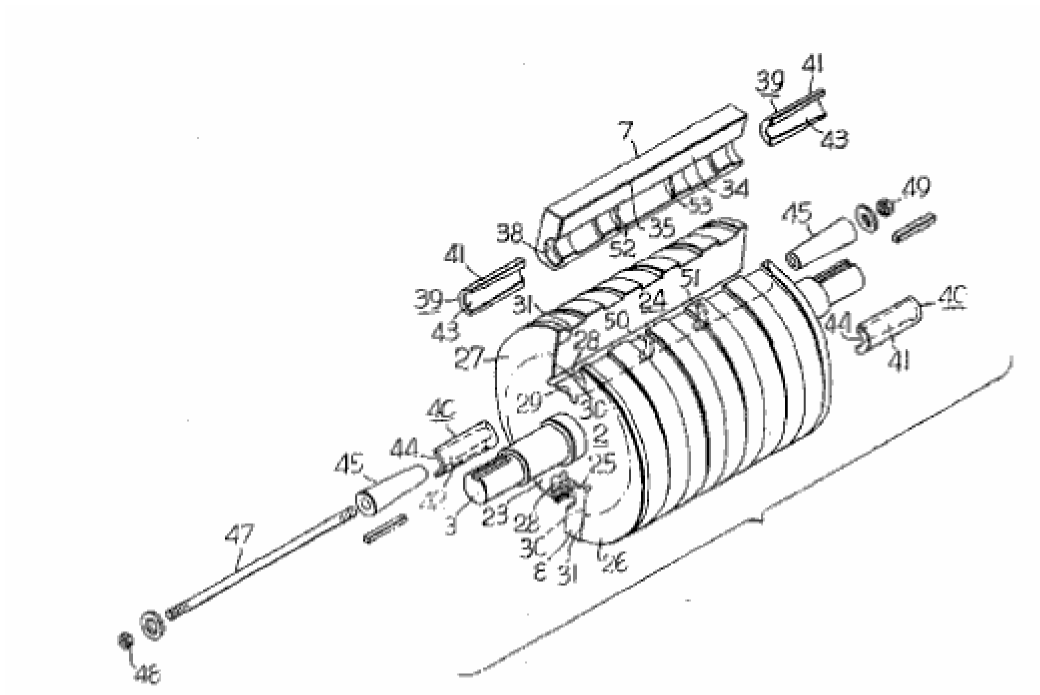


Fig 1.

Fig. 1 of the drawings is a view in elevation and partly in section, showing an impact crusher according to the present invention.

Fig. 2 is an exploded isometric view of a rotor for the crusher shown in Fig.1.

Referring to fig. 1 an impact crusher is shown which comprises a housing.

1. having disposed within a lower area of a rotor.
2. mounted on a shaft
3. which is carried by suitable journal bearing.
4. the housing 1 defines a material feed

Opening 5 over a feed chute 6 inclined downwardly towards the rotor 2. The feed chute 6 delivers rock to hammers 7 & 8 which are carried by 2 in a manner chute 6 directs feed rock to rotor 2 at a location where its hammers 7 & 8 are ascending with the result that the impact of hammers 7 & 8 on rock breaks the rock into smaller particles w3hich are thrown upwardly to break into even smaller particles upon impact with a complement of primary target breaker bars 12 and 13 which are carried by the sassing 1. a secondary crushing occurs when such particles drop downwardly from bars 12 and 13 to be again struck by hammers 7 and 8 and thrown towards a discharge are 14 where the particles impact with a vertical array of secondary target bars 15 d\close to the periphery of the rotor 2 one or more adjustable and yieldable breaker bars 16, 16' may be arranged. Adjustable and yieldable moan such as for bars 16 and 16' are well known. Any particles not between the bars 15 and into discharge area 14 progress downwardly towards the bar 16' where such particles are subjected to a final crushing as the particles are nipped and urged through the space between the rotor 2' and bar 16' to the lowest portion of the discharge area. The casing 1 may include a pivoted position 18 connected tro base structure 19 by a hinge so operative to open the casing and provided access to the internal mechanisms.

The rotor 2, shown in elevation in Fig-1 and the exploded isometric in Fig-2 will now be described. The rotor may be thought of as comprising an elongated central body portion 23 with a pari of axially extending slots 24, 25 and a pair of diametrically apposed spiral body extensions 26, 27. with reference to Fig.2 slot 24 is shown as having a leading wall 28, a floor 29 and a back wall 30. The body extensions 126, 27 each project progressively farther radially outward beginning at leading wall 28 of one slot and reaching maximum radial projection at a terminus defining a face 31 planner with back wall 30 of the other slot.

First and second hammer means, whose in both Fig-1 and Fig-2 as the hammer bars 7 & 8, are each arranged in one of the slots 24, 25 and project outwardly of such slots with a leading hammer face 34 extending outwardly and terminating with an edge 35 outward of slot edge 25 as far as the readially outer edge of the face 31.

Referring now again to Fig-2, an radial inner surface portion 38 of the leading hammer face 34 of bar 7, is shaped to define a concave semi cylindrical cavity. The leading slot wall 28 is also shaped to define a concave section in the cylindrical cavity 28, 38 at each end there of. The two bushing pieces 39, 40 are each halves of a complete bushing and

each have a convex semi cylindrical outer surface 41, 42 respectively, which together define a frusto conical cavity. A frustoconical plug 45 is provided for each bushing 39, 40 and each plyg 45 conforms to the shape of the frusto conical cavity defined by bushing surfaces 43, 44. each plug 45 has a central bore 46 through which a rod 47 may be inserted. The rod 47 is threaded on both ends for engagement with threaded nuts 48,49.

In the assembly of rotor 2, hammer 7 for example is placed is slot 24 on floor 29 and abutting against back wall 30, with wall 38 and surface 38, thereby co-operating to define there between a generally cylindrical cavity. One of the two piece bushings 39, 40 is inserted from and of the rotor 2 with apex ends of the inner frusto conical cavities 43, 44 pointed inwardly of the adjacent rotor ends. The bushings 39, 40 are moved inwardly untill the pieces abut against semi annular retaining walls 52, 53 in the cavity defined by surface 38. The retaining walls 50-53 limit movement of the bushings inwardly their adjacent rotor ends. With the two piece bushings 39, 40 each abutting against a pair of retaining walls 50, 52 and 51, 53 respectively, the bushing surface 41 is aligned to engage slot wall surface 41 is aligned to engaged slot wall surface 28, preferably with a diametrical plane x – x' along which the bushing is split into halves 39, 40 being substantially parallel to the adjacent slot back wall 30 referri9ng again to Fig.2, the frusto conical plugs as are aligned with their apex ends pointed inwardly and each is inserted into one of the bushing cavities defined by surface 43, 44. the rod 47 is then inserted through the bares of plugs 45 and the nuts 48, 49 are secured to its ends.

In the operation of the described assembly to source a hammer, such as hammer of as the plugs 45 are driven in their respective bushings 39, 40 the plugs 45 are moved inwardly toward each other, the bushing halves 39, 40 surrounding each plug 45 move apart. Finally, as hushing halves 39, 40 move apart, their convex semi cylindrical outer surface 41, 42 move apart and apply forces which are directed radially of the bushing halves 39, 40 to both push hammer 7 tangentially w.r.t the central body portion 23 of rotor 2, against the back wall 30 of slot 24, to securely lock the hammer 7 in its slot 24.

CHAPTER 5

STUDY OF FEEDER MECHANISMS

Feeder Mechanisms

Various conveyer systems

1. **Telescopic belt conveyor**

   They are designed to achieve high handling rates. Their dimensions can be altered to befit distinct machine requirements.

   Advantages

   → Reducing loading and unloading time

   → Improved handling efficiency

   →Improved operative safety

2. **Screw conveyor**

   The can be mounted in horizontal vertical and inclined configurations. These primarily consist of a conveyor screw, rotating in a stationary trough Appl's.

   → Can be used as feeders in crushers

   → Can be used as a vertical lifter for powder and granule.

3. **Vibrating Screen**

   They consist of robust vibrating s feens used widely for grading and sieving. Composing a main screen, screen web electric motor, eccentric block, rubber spring and coupler these vibrating screens can be customized to suit the customers requirement.

   Features

   → highly reliable and durable

   → Eccentric type systems

   → high screening capacity

   → No transmission of screen panels

   → Rigid and vibrating

   Appln's

   → Used in industries like crushing plants

CASING

The mill case is welded steel construction and built in three sections. The lower half is in one piece and upper half is in two sections. The feed intake section of the upper half is bolted to the lower half resulting in a permanent dust type connection between the feeding and mill intake.

The remaining top section is hinged for access to interior of the mills for changing hammers, hammer pins and screens. All mating surfaces are mechanized for an accurate, dust tight fit. Single latch door for easy maintenance and cleaning. Gasketed door for dust tight operation.

Screen

The degree of fineness of the ground product depends upon the size of the screen perforations. The screens are in two sections. In addition to the lower half circles screen. Mills have an extra quarter screen located in the hinged section of the top case. This increases the screen area by approximately 50% more than the conventional designs.

Quick screen Release

The lower screen is held securely in place by a quick screen release mechanism, ensuring a fight screen fit to guard against leakage of fine dust, which it allows quick and easy changing of screens.

CHAPTER 6

COMPUTERISATION

Hammer Design Program:

```
#include<stdio.h>
#include<conio.h>
#include<math.h>
void main()
float A,A’,T,D,p,m’,s,i,c,str;
const float F=12.3;
printf(“ENTER THE VALUE OF OPEN SCREEN AREA”);
scanf(“%f”,&A);
printf(”ENTER THE VALUE OF TOTAL TONNAGE PER HAMMER”);
scanf(%f”,&T);
printf(“ENTER THE VALUE OF EI”);
scanf(“%f”,&d);
printf(“ENTER THE VALUE OF I AND C”);
scanf(“%f%f”&i,&c);
printf(“ENTER THE VALUE OF STRESS ALLOWABLE”);
scanf(“%f”,&str);
printf(“THE MATERIAL SELECTED FOR HAMMER IS MS STEEL”);
a=48*E*I/2*l*l*l;
b=T*30.5;
c=T;
m=b*b-4*a*c;
x=-b+sqrt(m);
p=48*d*x/l*l*l;
m’=p*l/4;
s=m’*c/I;
if(s<str)
printf(“safe design”);
else
printf(“unsafe”);
}
```

Belt design program:

```
# include < studio. h>
# include < conio. h>
# include < math. h>
void main ( )
{ float P, β, d , w, T, μ, p,a,v,Tc, c, θ, k, N
char s [] = “Material selected for the belt is leather”
const float M= 0.15
const float allow _stress =7;
const float p = 1200;
Which ( 1)
{ printf (“enter the value of max power”)
scant (“% f “ &p);
printf (“enter the value of the belt angle”);
scant (“% f “ & β);
print f (“enter the value for dia of belt”);
scant (“% f” & d);
print (“enter the value of width of belt”);
scant (“% f, & w);
printf (“enter the value of thickness of belt”);
scant (“% f”, & 7);
printf(“enter the value of mass / length of the belt”);
scant (“% f”, & n);
prinft (“enter the value of r.p.m”);
scant (“% f” , & N);
b= tan (β/2) /(w/2);
p= tan (β/2);
a=1/2 * b*p;
v=3.14 * d* N/60;
Tc= m* (V*V);
```

C=0.55 * (D+d) +T;

Printf (“enter the new approx value of C from the table”);

Scant (“% f “ , & C);

θ=2* cos-1 (D-d)/2*c;

K= μ*θ;

$T_1 = T_1$;

$T_2 = T_2$;

$T = T_1 - T_c$;

σ = T/a;

If ($\sigma < \sigma_{all}$).

{ printf (“design is safe”);

break;

}

}

CONCLUSION

Impact crushers are the latest breed of crushers in use.They have proved to be more efficient than the other two major types of crushers and are rapidly replacing them.In the present design we have tried to concentrate on the design of major components of an Impact crusher and have been able to find certain results.

Computer aided technique is a rapidly growing method for any analysis, design and investigation purpose.But the constraint associated with this is that it requires sufficient accuracy in source codes. Source codes are of primary importance in any computer aided job.In the present project work, the source codes are written in C, but it can be done in other computer languages.

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